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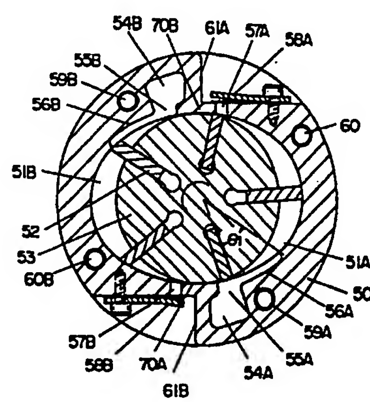
(54) ## ROTARY COMPRESSOR.

(57) A compressor which has a rotor (53) slidably provided with vanes (52); slidable vanes (52) provided in the rotor (53); a non-circular cylinder (50) containing the rotor (53) therein; side plates attached to the two side surfaces of the cylinder (50) to seal the sides surfaces of blade chambers (51-A), (51-B) formed by the vanes (52), the rotor (53) and the cylinder (50); suction holdes (56-A), (56-B); and discharge holes (57-A), (57-B). This construction suppresses the freezing capacity during high-speed driving by utilizing the suction loss when the pressure of a blade chamber is reduced to below the pressure of a coolant supply source during a suction stroke, the configuration is designed to vary in at least two stages so that the effective area of the passage from the suction hole within the blade chamber in the second half of the suction stroke is smaller than that in the first half, thereby obtaining an effective suppression effect on the freezing capacity during high-speed driving while maintaining low torque at low speeds and a high volumetric efficiency.

FIG.3



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SPECIFICATION

TITLE OF THE INVENTION

Rotary Compressor

TECHNICAL FIELD

This invention relates to a rotary compressor for car air conditioning which has, for example, vanes and changes the number of rotation in a wide range.

BACKGROUND ART

Generally, a sliding vane type compressor, as shown in Fig. 1, comprises a cylinder 1 having therein a cylindrical space, the side surfaces (not shown in Fig. 1) fixed to both sides of the cylinder 1 and sealing blade chambers 2a and 2b of the inner space in the cylinder 1, a rotor 3 disposed at the center thereof, and vanes 5 slidably engageable with grooves 4 provided at the rotor 3, suction bores 6a and 6b, being formed at the cylinder 1, discharge bores 7a and 7b being formed at the same, communication conduits 8a and 8b communicating with the blade chambers 2a and 2b formed in the cylinder 1, and set screws 9a and 9b at the suction side and those 10a and 10b at the discharge side being provided.

The vanes 5 project outwardly by a centrifugal force as the rotor 3 rotates so that the utmost ends of vanes 5 slidably move along the inner periphery of cylinder 1, thereby prevent leakage of gas from the compressor.

Fig. 2 is a sectional side view of the compressor, in which reference numeral 11 designates a front plate of side plate, 12 designates a rear plate, 13 designates a front casing, 14 designates a rotary shaft, 15 designates a shell, 16 designates an annular suction conduit formed between the front casing 13 and the front plate 11, 17 designates a suction piping joint, 18 designates a suction conduit shown by the chain line, 19 designates a disc for clutch means, and 20 designate a pulley for clutch means.

The compressor, as shown in Fig. 1, having the cylinder 1 not-circular in the inner surface in section requires a plurality of pairs of suction bores and discharge bores.

The compressor having a cylinder of the inner surface about elliptic in section discharges a refrigerant compressed in the right-hand and left-hand blade chambers 2a and 2b through two discharge bores 7a and 7b into a common space formed of cylinder 1 and shell 15.

Supply of the sucked refrigerant into two blade chambers 2a and 2b is separate from the discharge side and cut off therefrom by use of a construction shown in Fig. 2.

In detail, between the front plate 11 and the front casing 13 is formed the annular suction conduit 16

communicating in common with two suction bores 6a and 6b and the piping joint 17 provided at the front casing 13 connects the conduit 16 with an external refrigerant supply source (an exit of an evaporator).

Such construction need only provide each one suction and piping joint even in a multirobe type compressor having two or more cylinder chambers.

Such sliding vane type rotary compressor can be small-sized and simple in construction rather than the reciprocating compressor complex in construction and of many parts, thereby having recently been used for the car cooler compressor. The rotary compressor, however, has the following problems in comparison with the reciprocating compressor.

In other words, in a case of car cooler, a driving force is transmitted from an engine to a pulley 20 at a clutch means through a belt to drive a rotary shaft of the compressor. Hence, when the sliding vane type compressor is used, its refrigerating capacity rises about linearly in proportion to the number of rotations of the car engine.

On the other hand, in a case of using the reciprocating compressor, the follow-up property of a suction valve becomes poor during the high speed rotation and a compressed gas cannot be fully sucked into the cylinder.

As a result, the refrigerating capacity leads to saturation during car driving at high speed. In brief, while the reciprocating compressor automatically suppresses the refrigerating capacity during the high speed driving, the rotary one acts not so and deteriorates its efficiency as the compression work increases, or is conditioned in subcooling (excessive cooling). In order to solve the above problem, the method has hitherto been proposed that a control valve for changing an opening area of communicating conduit is provided at the conduit communicating with the suction bores 6a and 6b at the rotary compressor, the opening area being restricted during the high speed rotation to utilize the suction loss for performing capacity control. In this case, however, the control valve should extra be attached, thereby having created the problem in that the compressor is complex in construction and expensive to produce. Another method, which uses a fluid clutch or planetary gears not to increase the number of rotations more than the predetermined value has hitherto been proposed for eliminating the excessive capacity of compressor during the high speed driving.

However, the former method is larger in energy loss caused by friction heating on the relative-moving surface and the latter is added with a planetary gear mechanism of many parts to be larger in size and configuration,

thereby being difficult to put in practical use because the tendency of energy saving recently increasingly requires simplification and miniaturization of compressor.

After the detailed research by the inventors of transient phenomena of pressure in the blade chamber in a case of using the rotary compressor for the purpose of solving the aforesaid problem created in accompaniment with the refrigeration cycle for a car cooler in a rotary system, it has been formed that even when the rotary compressor is used, parameters for the suction bore area, discharge amount, and the number of vanes, are properly selected and combined, whereby the self-suppression acts effectively on the refrigerating capacity during the high speed rotation as the same as in the conventional reciprocating compressor, which has been proposed in the specification of Japanese Patent Application No. Sho 55-134048.

Also, after study of general characteristic of compressor in consideration of power consumption as well as the volumetric efficiency, it has been found that the effective suction area is allowed to vary in at least two stages and the effective areas in the first half and the second half in the suction stroke are properly set so that during the low speed rotation a drive torque is expected to decrease and moreover during the high speed

rotation a sufficient capacity control effect is obtained, which has been proposed in the specification of Japanese Patent Application No. Sho 56-62875.

DISCLOSURE OF THE INVENTION

This invention has expanded application of the above to a general compressor. For example, this invention has designed a concrete construction of compressor comprising a not-circular cylinder when subjected to capacity control. An object of the invention is to provide a compressor having two laterally symmetrical chambers (two robes) in a space formed by a rotor and an elliptic cylinder, providing at least four or more vanes disposed separately within the rotor, and forming the suction ports and suction grooves so that the effective suction area changes in about two stages during the suction stroke, thereby operating the compressor with low torque without lowering the refrigerating capacity during the low speed driving and obtaining an effective suppression effect during the high speed driving. The compressor of the invention comprises a rotor, vanes contained slidably therein, a not-circular cylinder containing therein the rotor, side plates fixed to both sides of cylinder and sealing spaces in blade chambers formed of the vanes, rotor and cylinder at both sides of blade chamber, suction bores, and discharge bores, thereby utilizing a suction loss caused by

pressure within the blade chamber lower than that of refrigerant supply source during the suction stroke so as to suppress the refrigerating capacity of the compressor during the high speed driving, and is characterized in that an effective area of each passage from the suction bore to the blade chamber is adapted to change in at least two stages to thereby be made smaller in the second half of the suction stroke than in the first half of the same.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a sectional front view of a conventional sliding vane type rotary compressor,

Fig. 2 is a side view of the compressor in Fig. 1,

Fig. 3 is a sectional front view of an embodiment of a rotary compressor of the invention,

Fig. 4-(a) is a view showing the positional relation between vanes and rotor of the compressor in Fig. 3 during the suction stroke,

Fig. 4-(b) is a view showing the positions of vanes and rotor of the same just before a termination of the suction stroke,

Fig. 4-(c) is a view showing the positional relation between the respective vanes and the rotor at the termination of suction stroke,

Fig. 5 is a sectional view of a suction groove,

Fig. 6 is a sectional front view of a compressor with three vanes,

Fig. 7-(a) is a sectional front view of a compressor with four vanes working during the suction stroke,

Fig. 7-(b) is a view showing the positions of vanes and rotor of the four vane rotary compressor at a terminate of suction stroke,

Fig. 8 shows a pattern of the number of vanes and effective suction area,

Fig. 9 shows a relation between the effective suction area and a travelling angle of each vane,

Figs. 10, 11 and 12 show the relations between the pressure in a blade chamber and the travelling angle of the respective vanes,

Fig. 13 shows the pressure drop rate with respect to the number of rotations of the rotor,

Fig. 14 is a model chart of pressure-volume curves,

Fig. 15 is a model chart of PV curves in the embodiment of the invention,

Fig. 16 shows torque with respect to the number of rotations of rotor,

Fig. 17 shows the suction loss to the number of rotations of rotor,

Fig. 18 shows an excessive compression loss to the number of rotations of rotor,

Fig. 19 shows the pressure drop rate with respect to the number of rotations of the rotor when the effective area in the second half of suction stroke 1,

Fig. 20 is a model chart of pressure drop rate with respect to the number of rotations of rotor,

Fig. 21 is a graph showing the pressure drop rate with respect to the number of rotations when the effective suction area is constant, and

Fig. 22 is a sectional view of a modified embodiment of the invention.

BEST MODE FOR CARRYING OUT THE INVENTION

Next, the invention will be described in the following order:

I Basic construction of the invention

II Principle of the same

III Modified embodiment of the same

(I) Explanation on the basic construction of the invention

Next, explanation will be given on an embodiment of applying this invention to a two robe type (of about elliptic cylinder) sliding vane compressor.

Fig. 3 is a sectional front view of an embodiment of a compressor of the invention, in which reference numeral 50 designates a cylinder, 51A designates a blade chamber A, 51B designates a blade chamber B, 52 designates vanes disposed into a rotor 53 spaced circumferentially thereof

at five equal intervals, 54A and 54B designate suction bores, 55A and 55B designate suction nozzles, 56A and 56B designate suction grooves formed at the inner periphery of cylinder 50, 57A and 57B designate discharge bores, 58A and 58B designate discharge valve holders, 59A and 59B designate fixing bolts at the suction side, 60A and 60B designate fixing bolts at the discharge side, and 61A and 61B designate cutouts formed at the positions where the suction side and discharge side are separate laterally from each other.

Now, the embodiment of the compressor of the invention in Fig. 3 is different largely from the conventional compressor (in Fig. 1) in the following points:

(i) The compressor in Fig. 3 form the suction bores 54A and 54B in proximity to the top portions 70A and 70B of cylinder 1.

(ii) The fixing bolts 59A and 59B for fixing the cylinder 50 with the front plate and rear plate (not shown but in Fig. 2) are disposed ahead of suction bores 54A and 54B in the rotating direction of rotor 50.

(iii) At the inner surface of cylinder 50 are provided suction grooves 56A and 56B elongate across an angle of $\theta 1$.

A sliding vane compressor comprising a cylinder other than the round one is to be hereinafter called the multirobe type compressor.

Table 1

Parameter	Reference	Number of vanes			Round cylinder of two vanes for reference
		3	4	5	
Rotary angle of vane end at termination of suction	θ_s	150°	135°	126°	270°
Travelling angle of cylinder groove (control zone)	$\theta 1$	8.6°	23.4°	32.6°	70°
Ratio of θ_1 to θ_s	θ/θ_s	0.057	0.173	0.259	0.259

In the above Table 1, the rotary angle: θ_s of vane end at the termination of suction, travelling angle: θ_1 of cylinder groove, and port position angle: θ_2, are defined as follows:

In Fig. 4, reference numeral 62a designates a blade chamber at the down-stream side, 60b designates a blade chamber at the upstream side, 70A designates a top portion of cylinder 50, 64a designates a vane a, 64b designates a vane b, and 65 designates an end of suction groove 56A.

The position where the vane end passes along the top portion 70A around the axis of rotation of rotor 53, is represented by $\theta = 0$, in which $\theta = 0$ is made the origin and an angle at the optional position of vane end is represented by θ. When the downstream side blade chamber 62a is viewed, Fig. 4-(a) shows the vane 64a having already passed the suction bore 54A and suction groove 56A and

is positioned at an angle of about $\theta = 90°$, a refrigerant being supplied from the suction bores 54A directly to the downstream side blade chamber 62a as shown by the arrow.

Fig. 4-(b) shows a condition just before the termination of suction stroke, in which the refrigerant is supplied to the downstream side blade chamber 62a from between the vane 64b and the suction groove 56A.

Fig. 4-(c) shows a condition of termination of suction stroke of downstream side blade chamber 62a (where $\theta = \theta_S$), in which the utmost end of vane 64b is positioned at the suction groove end 65. At this time, the downstream side blade chamber 62a partitioned by the vanes 64a and 64b is maximum in volume.

The port position angle θ_2 represents an angle between the top portion 70A at the cylinder 50 and the center of suction port 54A, the travelling angle θ_1 of the cylinder groove in the control zone representing an angle of travelling of vane 64b along the suction 56A until the suction stroke terminates.

In the embodiment, the suction ports 56A and 54B are positioned at the centers thereof at an angle of $\theta_1 = 21.4°$.

The closer the suction port is positioned to the top portion (where $\theta = 0$), the smaller a gap between the rotor 53 and the cylinder 50 becomes, whereby the effective suction area is difficult to enlarge. Hence, it is

necessary for the travelling angle of θ = 20 to 30° or more in general to form the suction port apart from the top portion 70A.

Fig. 5 is a sectional view of suction groove 56A formed at the cylinder 50, in which the effective area of suction groove is of a product of sectional area s = e x f of suction groove 56A multiplied by the coefficient of contraction.

Now, in the embodiment of the invention, the multi-robe type compressor is used to change step the effective suction area during the suction stroke, thereby having enabled realization of the compressor which is operable at low speed, is less in volumetric efficiency loss, saves power consumption, and has an effective suppression effect on the refrigerating capacity during the high speed driving only.

The multi-robe type compressor is smaller in total weight of refrigerant allotted to one blade chamber in comparison with the compressor of round cylinder, thereby being advantageous in the high speed durability with respect to fluid compression or excess compression. It will be detailed in Item (II) why the stepped change of suction area makes effective the capacity control characteristic, but nextly, the compressor of multi-robe type of three vanes and four vanes will be compared with that of

the aforesaid five vanes in the following description.

Fig. 6 shows a construction of the three vane compressor, in which reference numeral 100 designates a rotor, 101 designates a cylinder, 102 designates a suction port, 103 designates a vane a, 104 designates a vane b, and 105 designates a blade chamber A. A travelling angle θ_1 of vane b 104 following the vane a 103 is only 8.6° with respect to the cylinder groove, thereby being difficult to construct the effective suction area in a stepped manner during the suction stroke.

Fig. 7 shows a construction of the four vane compressor, in which reference numeral 200 designates a rotor, 201 designates a cylinder, 202 designates a suction port, 202a designates a vane a, 203 designates a vane b, and 204 designates a blade chamber A.

In this case, the above travelling angle is as θ_1 = 23.4° and θ_1/θ_s = 0.173, which is slightly disadvantageous in the stepped construction in comparison with the five vane compressor of θ_1/θ_s = 0.259 or the two vane compressor of round cylinder (wherein θ_2 = 20°).

Fig. 8 shows a pattern of effective suction area obtainable by the respective compressors different in numbers of vanes. In a case of applying the capacity control to the multi-robe type compressure, it is seen from the above that the number of vanes should be properly

selected for reducing torque and improving the Cop (control processor) especially in the low speed zone.

The embodiment in Fig. 3, in comparison with the conventional construction in Fig. 1, enabled the travelling angle θ_1 to the cylinder groove to be enlarged sufficinetly from a design of arrangement of suction bores 54A and 54B and suction grooves 56A and 56B as described in the aforesaid items (i) to (iii).

In the conventional construction in Fig. 1, it is difficult to set the effective suction area in the patterns (b) to (f) as shown in Fig. 9.

(II) Explanation of Principle of the Invention

Next, it will be explained together with a result of analysis of characteristic in the suction stroke why the stepped change of effective suction area during the suction stroke is effective.

Fig. 9 and Table 2, show in the patterns (a) to (f) the effective suction area a with respect to the travelling angle of vane, where the effective suction area has been arranged by the capacity control parameter K_2 in order to carry out relative comparison of characteristics of various compressors (K_2 is to be discussed below).

Table 2

Pattern	Effective area arranged by $K_2 = a\ _s/V_o$	
	First half: $K_{21} = a_1\theta_s/V_o$	Second half: $K_{22} = a_2\theta_s/V_o$
(a)	0.0451	
(b)	0.050	0.0436
(c)	0.060	0.0421
(d)	0.080	0.0411
(e)	0.100	0.0391
(d)	0.140	0.0386

In the pattern (a), the effective suction area a is always constant during the suction stroke, which is realized by constructing the compressor to make larger the sectional area: $S = 2 \times e \times f$ of suction groove 56A with respect to an area of suction bore 54A (see Fig. 5).

The patterns (b) to (f) shows the effective suction area made larger in the first half of suction stroke and smaller in the second half ofthe same. Especially, the patterns (b) to (g) corresponds to the present invention aiming at reducing torque during the low speed driving.

In the embodiment, reversely to the pattern (a), the effective areas of suction grooves 56A and 56B were made smaller than those of suction bores 54A and 54B.

Next, explanation will be given on the characteristic analysis carried out to catch in detail the transient

phenomenon of refrigerant pressure, which is an important point for the invention.

The transient characteristic of pressure in the blade chamber is given by the following energy equation:

$$\frac{Cp}{A}GT_A - Pa\frac{dVa}{dt} + \frac{dQ}{dt} = \frac{d}{dt}\left(\frac{Cv}{A}\gamma_a VaTa\right) \quad .... (1),$$

where G: mass flow of refrigerant, Va: blade chamber volume, A: thermal equivalent of work, Cp: specific heat at constant pressure, T_A: refrigerant temperature at the supply side, k: ratio of specific heat, R: gas constant, Cv: specific heat in constant volume, Pa: pressure in blade chamber, Q: quantity of heat, γa: specific weight of refrigerant in blade chamber, and Ta: refrigerant temperature in blade chamber. In addition, in the following equations (2) to (4), a: effective area of suction bore, g: gravitational acceleration, γ_A: specific weight of refrigerant at the supply side, and Ps: refrigerant pressure at the supply side.

In the equation 1, the first term at the left side represents the thermal energy of refrigerant taken into the blade chamber through the suction bore at the unit time, the second term at the same represents work of refrigerant pressure with respect to the exterior at the unit time, the third term at the same represents thermal energy flowing into the blade chamber from the exterior through the outer wall, and the right side represents

an increment in the internal energy of system at the unit time. Assuming that the refrigerant conforms with the rule of ideal gas and the suction stroke is rapid to cause the adiabatic change, from $\gamma a = Pa/RTa$ and $\frac{dQ}{dt} = 0$, the following equation is given:

$$G = \frac{dVa}{dt}\left(\frac{A}{CpT_A} + \frac{1}{kRT_A}\right)Pa + \frac{Va}{kRT_A} \cdot \frac{dPa}{dt} \quad \ldots\ldots\ldots\ldots (2)$$

Also, by use of relational expression of $\frac{1}{R} = \frac{A}{Cp} + \frac{1}{kR}$,

$$G = \frac{1}{RT_A} \cdot \frac{dVa}{dt} \cdot Pa + \frac{Va}{kRT_A} \frac{dPa}{dt} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

is obtained.

A mass flow of refrigerant passing through the suction bore is applicable with the theory of nozzle, whereby the equation:

$$G = a\sqrt{2g\gamma_A Ps\frac{k}{k-1}\left[\left(\frac{Pa}{Ps}\right)^{\frac{2}{k}} - \left(\frac{Pa}{Ps}\right)^{\frac{k+1}{k}}\right]} \quad \ldots\ldots\ldots\ldots (4)$$

is obtained. Therefore, the equations (3) and (4) are solved to obtain the transient characteristic of pressure Pa in the blade chamber.

Fig. 10 shows the transient characteristics of pressure in the blade chamber in a case of the effective suction area (c) in Fig. 9 obtained by using the number of rotations as the parameter.

The equations (3) to (4), five vanes in Table 1, conditions in Table 3, and initial conditions of t = 0 and Pa = Ps. Since the refrigerant in the refrigerating cycle

for the car cooler usually uses R12, the analysis was carried out by using $k = 1.13$, $\gamma_A = 16.8 \times 10^{-6} kg/cm^2$ and $T_A = 28.3°K$

In Fig. 10, the pressure in the blade chamber Pa during the low speed rotation (ω = 1000 rpm) and in the vicinity of $\theta/\theta_s = 1$ ($\theta = \theta_s = 126°$) of the termination of suction stroke reaches supply pressure Ps = 3.18 kg/cm^2 abs., thereby creating no loss in the pressure in the blading chamber at the termination of suction stroke. Upon increasing the number of rotations, the refrigerant supply cannot catch the volume change of blade chamber so that the pressure loss at the termination of suction stroke ($\theta/\theta s = 1$) increases. For example, when N = 5000 rpm is given, the pressure loss $\Delta P = 1.30 kg/cm^2$ (Pa/Ps = 0.591) is created with respect to the supply pressure Ps to cause a decrease in the gross weight of sucked refrigerant and lead to large reduction of refrigerating capacity.

Table 3

Parameters	Reference	Embodiments
Refrigerant pressure at supply side	Ps	3.18 kg/cm^2 abs
Refrigerant temperature at supply side	T_A	283°K
Refrigerant pressure at discharge side	Pd	15.51 kg/cm^2 abs
Number of rotations	N	600 - 5000 rpm

The effective suction areas in Fig. 9-(f) and that in Fig. 9-(a) are shown in Figs. 11 and 12 respectively.

Now, when the pressure in the blade chamber at the termination of suction stroke is represented by Pa = Pas, the pressure drop rate: ηp is defined as follows:

$$\eta p = (1 - \frac{Pas}{Ps}) \times 100 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

Fig. 13 is a graph showing a characteristic of the pressure drop rate with respect to the number of rotations when the effective suction areas are different respectively (in Figs. 9-(a) to -(f)). Namely,

1. During the low speed rotation of N = 2000 rpm, the compressors having the effective suction areas of (a) to (f) in Fig. 13 are about identical in pressure drop rate with each other.

2. During the high speed rotation of N = 5000 rpm, the compressor of (a) whose effective suction area is constant during the suction stroke, is the maximum in pressure drop rate.

3. The embodiment in Table 1 of effective suction area in Fig. 13-(c), has the characteristic corresponding about to the above (c), the compressor of the same in (f) having a considerably smaller ηp of the effect of capacity contro.

The pressure drop rate may be considered to be about equal to that for the gross weight of refrigerant filled in the blade chamber at the termination of suction stroke.

Accordingly, the compressor having the pressure drop rate with respect to the number of rotations of the characteristic as shown in Fig. 13-(c), even when viewed from the control amount only of refrigerant, is known to obtain the refrigerating capacity nearly conforming to the ideal one as follows:

i During the low speed rotation, the suction loss lowers the refrigerating capacity a little.

The reciprocating compressor of self suppression effect for the refrigerating capacity is characterized in that its suction loss is minimum at low speed rotation, but the rotary compressor of the invention has the characteristic not inferior to the reciprocating one.

ii During the high speed rotation, the rotary compressor obtains the refrigerating capacity suppressing effect equal to or more than that of conventional reciprocating compressor.

iii In a case of raising the number of rotations to more than 1800 - 2000 rpm, the suppression effect is obtained so that, when used as the compressor for the car cooler, the refrigerating cycle of ideal energy saving and in good feeling has been materialized.

iv The drive torque lowers about in proportion to the number of rotations, thereby having obtained the effect of large energy saving during the low and high speed

rotations.

The effects described in Items i to iii have already been disclosed in the Japanese Patent Application No. Sho 55-134048.

The embodiment of the present invention is characterized, besides the above effects in Items i to iii, in that the multi-robe type compressor of not-circular cylinder, even when used, can obtain lower power consumption at the low speed rotation.

Now, in a case of applying the capacity control, the drive torque of compressor includes the following items:

1. A loss in the suction stroke.
2. Compression power at the compression stroke.
3. A loss by excessive compression pressure.

The above items 1 to 3 will be explained according to the Figs. 14 and 15 of preferable model.

In Fig. 14, a curve N_1 described by a, b, c and d shows a standard polytropic suction compression stroke. Also, a curve N_2 described a, b', e, g and d applies the capacity control, the curves N_1 and N_2 showing the effective suction area constant during the suction stroke, for example, the PV chart of effective area in Fig. 9-(a). In a case of applying the capacity control, the pressure Pa in the blade chamber at the beginning point of compression

stroke lowers as the number of rotations increases. In a case of not applying the capacity control, since the refrigerant is filled completely into the blade chamber, the pressure Pa in the blade chamber at the compression stroke starting point b, i.e., Va = Va max. (or the suction stroke termination) is constant regardless of the number of rotations.

Referring to Fig. 15, a curve N_3 corresponds to the PV chart in Figs. 9-(b) to (f) where the effective suction area is two-stepped, in which an area S_1: power loss in the suction stroke, that S_2: decrement of compression power by the capacity control effect, and that S_3: loss of excessive compression power.

In a case where the effective suction area is constant in the suction stroke (in Fig. 9-(a)), since the pressure Pa in the blade chamber starts to lower when the volume Va of blade chamber is still small, its suction power loss S_1 (in Fig. 14) is larger. On the other hand, in a case where the effective suction area is larger in the first half of suction stroke and smaller in the second half of the same (for example, in Fig. 9-(c)), since a drop of pressure Pa in the blade chamber is smaller in the first half, the suction loss S_1 (in Fig. 14) as a whole becomes smaller in comparison with the former case. Fig. 16 shows an example of characteristic of drive torque with

respect to the number of rotations when the patterns of effective suction areas are different from each other.

Figs. 17 and 18 show the suction loss and excessive compression loss of the respective items (a) to (f) with respect to the number of rotations, from which it is seen that the smaller the effective suction area during the suction stroke is, the larger the suction loss becomes, and reversely the excessive compression loss becomes larger.

As seen from the above result, the effective suction area is made stepped to enable the rotor to rotate at low torque and low speed keeping moderate the capacity control effect. The stepped construction of effective suction area, as abovementioned, is difficult for the three vane type, whereby the embodiment of five vanes is the best.

Also, the embodiment of four to five vanes was proper because the number of vanes increased more than the need has increased a mechanical sliding loss between the vane and the cylinder.

Now, volume Va of blade chamber is the function of rotor diameter Rr or the cylinder configuration or the like, so that a method will be proposed which uses the following approximate functions to arrange the equations (3) and (4) to catch the correlation between the respective parameters and the capacity control effect.

The maximum suction volume of refrigerant is represented by Vo and $\psi = \Omega t = (\pi\omega/\theta_s)t$ is used to convert an angle θ to ψ, at which time ψ varies from 0 to π and $f(0) = 0$ and $f'(0) = 0$ at $t = 0$ are obtained. Also, the approximate function $f(\pi)$ of $f(\pi) = 1$ and $f'(\pi) = 0$ when the suction stroke terminates at $t = \theta_s/\omega$, is defined.

At this time, volume Va is given by $Va(\pi) \fallingdotseq Vo \cdot f(\psi)$ ...(6).

For example, given

$$f(\psi) = \frac{1}{2} \cdot (1 - \cos) \quad \ldots\ldots (7)$$

and $\eta = Pa/Ps$,

$$G = \frac{Ps\Omega Vo}{RT_A} \{f'(\psi) \cdot \eta + \frac{f(\psi)}{k} \cdot \frac{d\eta}{d\psi}\} \quad \ldots\ldots (8)$$

follows.

The equation (4) is arranged and

$$G = a\sqrt{Ps \cdot \gamma_A \cdot 2g \cdot \frac{k}{k-1}(\eta^{\frac{2}{k}} - \eta^{\frac{k+1}{k}})} \quad \ldots\ldots (9)$$

is obtained.

Accordingly, from the equations (8) and (9),

$$K_1 \cdot g(\eta) = f'(\psi) \cdot \eta + \frac{f(\psi)}{k} \frac{d\eta}{d\psi} \quad \ldots\ldots (10)$$

and

$$g(\eta) = \sqrt{\frac{k}{k-1} (\eta^{\frac{2}{k}} - \eta^{\frac{k+1}{k}})} \quad \ldots\ldots (11)$$

are obtained, where K_1 becomes the dimensionless quantity as follows:

$$K_1 = \frac{a\theta s}{Vo\pi\omega} \sqrt{2gRT_A} \quad \ldots\ldots (12).$$

In a case of sliding vane type compressor, when Vth is assumed to be a theoretical discharge amount, n the number of vanes, and m the number of robes, normally Vth = n x m x Vo is substituted in the equation (12), thus obtaining

$$K_1 = \frac{a\theta_s nm}{Vth\pi\omega}\sqrt{2gRT_A} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (13)$$

In addition, a ratio of specific heat in the equation (10) is the constant depending only on the kind of refrigerant.

In the equation 13, the effective suction area a is the function of vane travelling angle ψ of the dimensionless quantity, whereby the parameter K_1 also becomes the function of ψ.

Hence, the solution $\eta = \eta(\psi)$ of equation (10) is decided by a value of $K_1(\psi)$.

R and T_A in the equation (13) are set not by the construction of compressor, but under the same conditions, whereby the capacity control parameter can be re-defined as follows:

$$K_2(\psi) = 2\theta_s/Vo \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (14)$$

In other words, the characteristic of pressure in the blade chamber during the suction stroke is seen to be decided principally by the above $K_2(\psi)$. Here, K_{21} and K_{22} are defined as follows by use of the effective suction areas a_1 and a_2 in the first half of suction stroke and in

the second half of the same respectively:

$$K_{21} = \frac{a_1 \cdot \theta_s}{V_o} \quad \text{.................................} \quad (15)$$

$$K_{22} = \frac{a_2 \cdot \theta_s}{V_o} \quad \text{.................................} \quad (16).$$

After examination of Figs. 9 and 13, the following matters are known. In other words, when the effective area a_1 (or K_{21}) in the first half of suction stroke is largely changed, the pressure loss ηp during the high speed driving, but not so much during the low speed driving. For example, ηp when $N = 2000$ rpm can be made constant only by compensating to a minimum $(0.0386 < K_{22} < 0.0436)$ the effective area a_2 (or K_{22}) in the second half of suction stroke.

Next, in order to catch how the pressure drop rate ηp changes with respect to the number of rotations ω when the effective suction area in the second half of suction stroke is changed, analysis will be given on the following cases. Fig. 19 shows the characteristics of ηp with respect to N when the effective suction area a_2 (i.e. K_{22}) in the second half of suction stroke is changed under the respective conditions in Table 4 while keeping constant ($K_{21} = 0.060$) the effective suction area in the first half of suction stroke.

Table 4

	K_{22}
(g)	0.030
(h)	0.040
(i)	0.050
(j)	0.060

The above results are summarized by use of Fig. 20 model view as follows:

1. When K_{21} is changed, the slope of ηp with respect to the number of rotations N changes as A → C.

2. When K_{22} is changed, the curve of ηp with respect to N moves in parallel as A → B.

From the above, the effective area in the first half of suction stroke, in other words, the parameter K_{22} in the second half is included between (a) and (f) in a practical range as

$$K_{22} < K_{21} < 0.140 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (17).$$

When the effective suction area a is constant during the suction stroke, the parameter $K_1(\psi)$ obtained from the equation (13) becomes constant. When the effective suction area is constant, the following parameter K_2 is re-defined:

$$K_2 = \frac{a\theta s}{Vo} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (18).$$

In a case where the effective suction area during the suction stroke is constant, ΔT = 10 deg. is assumed as

superheat and under the condition of $T_A = 283°K$ the equations (3) and (4) are solved so that the results therefrom have been arranged by the parameter K_2 and shown in Fig. 21.

As apparent from comparison of Fig. 19 with Fig. 21, for the curve of K_{22} equal to K_2, in spite that the parameter K_{21} in the first half of suction stroke is different from K_2, the values of number N of rotations of $\eta p \approx 0$ are almost equal to each other. In brief, it is known that the number of rotations: Ns to start the capacity control is decided almost by the effective area a_2 (parameter K_{22}) regardless of the effective area a_1 (parameter K_{21}) in the first half (regarding Ns, see Fig. 20 of the model graph).

Now, the number of rotations of the car engine during the idling of car is normally set to N_1 = 800 to 1000 rpm.

Also, the number of rotations of the same when the car is running at the speed of u = 40 km/h, is N_2 = 1800 to 2200 rpm.

After research of application of the embodiment of the invention into usual cars, it was most desired to set the starting point of capacity control in a range of $N_1 < N_s < N_2$.

From Fig. 21, a range of parameter K_{22} is given in the following inequality:

$$0.025 < K_{22} < 0.055 \quad \text{........................} \quad (19) .$$

The effective suction areas a_1 and a_2 for computation of the equations (15) and (16) need only use the average values respectively.

In addition, the effective suction area is obtained from the product of sectional area depending on a geometric configuration of suction passage and coefficient of contraction.

As seen from the above, the embodiment of the compressor of the invention could be constructed to simultaneously satisfy the equations 17 and 19 and sufficiently obtain the capacity control in low torque during the low speed driving and also even at the high speed driving.

(III) A Modified Embodiment of the Invention

A modified embodiment of the invention is shown in Fig. 22, in which reference numeral 300 designates a rotor, 301 designates a cylinder, 302 designates vanes, 303 designates suction bores, 304 designates suction grooves, 305 designates set screws at the suction side, 306 designates set screws at the discharge side, and 307 designates suction nozzles.

In Fig. 22 construction, the set screws 305 at the suction side for fixing the front plate, rear plate (both are not shown) and cylinder 301, each were provided between the suction bore 303 and the top portion 308 of

cylinder, where each suction nozzle 307 was positioned at the center in proximity to the top portion 308 in order to sufficiently enlarge the travelling angle θ (see Table 1) of cylinder groove.

As seen from the above, the multi-robe type compressor having the effective suction area applied with the stepped change has been proposed of its construction. It is effective for leakage of refrigerant from the high pressure ride into the blade chamber during the suction stroke to enlarge the effective suction area in the first half, thereby largely contributing to an improvement in the volumetric efficiency during the low speed driving.

Industrial Applicability

As seen from the above, the present invention is summarized of its effect as follows:

1. Less refrigerating capacity loss at low speed rotation (1000 to 2000 rpm).

2. Large suppression effect on the refrigerating capacity obtained at high speed rotation (3500 to 5000 rpm).

3. Low torque drive especially during the low speed rotation.

The above items 1 to 3 are realizable by the present invention.

What is claimed is:

1. A rotary compressor comprising a rotor in which vanes are slidably provided, said slidable vanes provided in said rotor, a not-circular cylinder containing therein said rotor, side plates fixed to the two side surfaces of said cylinder and sealing spaces between the side surfaces of blade chambers formed of said vanes, rotor, and cylinder, and suction bores and discharge bores, so that a suction loss of pressure in said blade chamber lowering from that of a refrigerant supply source during the suction stroke is utilized to suppress high speed driving, said compressor characterized in that an effective area of a passage from each of said suction bores to said blade chamber is adapted to change in two steps so as to be smaller in the second half of the suction stroke than in the first half thereof.

2. A rotary compressor according to claim 1 characterized in that when a travelling angle of each of said vanes in the second half of suction stroke is represented by θ_1 and the whole travelling angles of vanes during the suction stroke are represented by θ_S, said vanes are constructed in a range of $\theta_1/\theta_S > 0.170$.

3. A rotary compressor according to claim 1 characterized in that when the maximum suction volume of the refrigerant is represented by Vo, the effective suction area in the

first half of the suction stroke is represented by a_1, and $K_{22} = a_1\theta_s/Vo$ is given, said compressor is constructed in a range of $0.025 < K_{22} < 0.055$.

FIG. 1

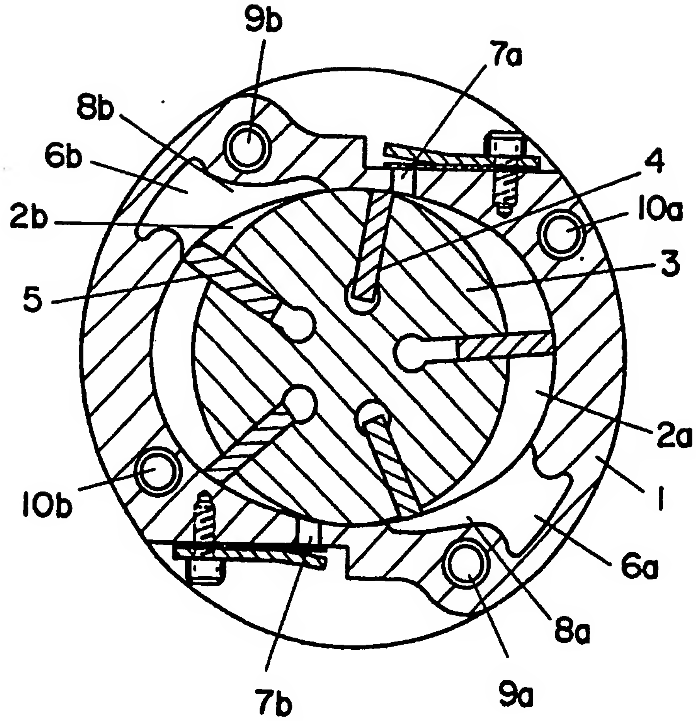


FIG.2

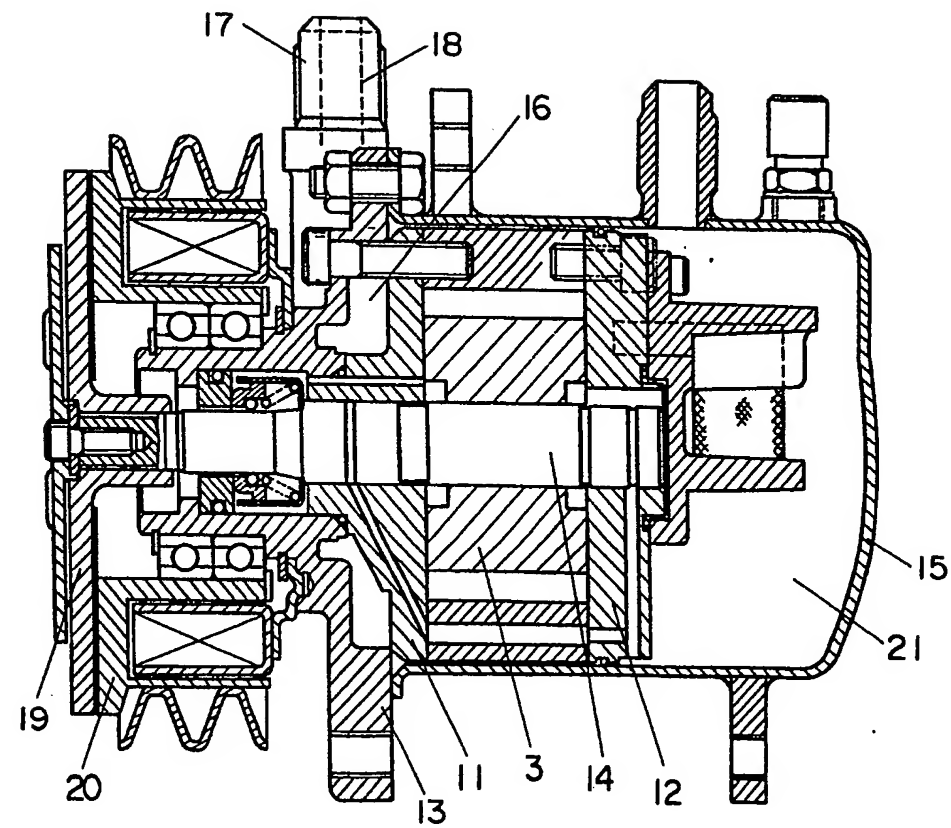


FIG. 3

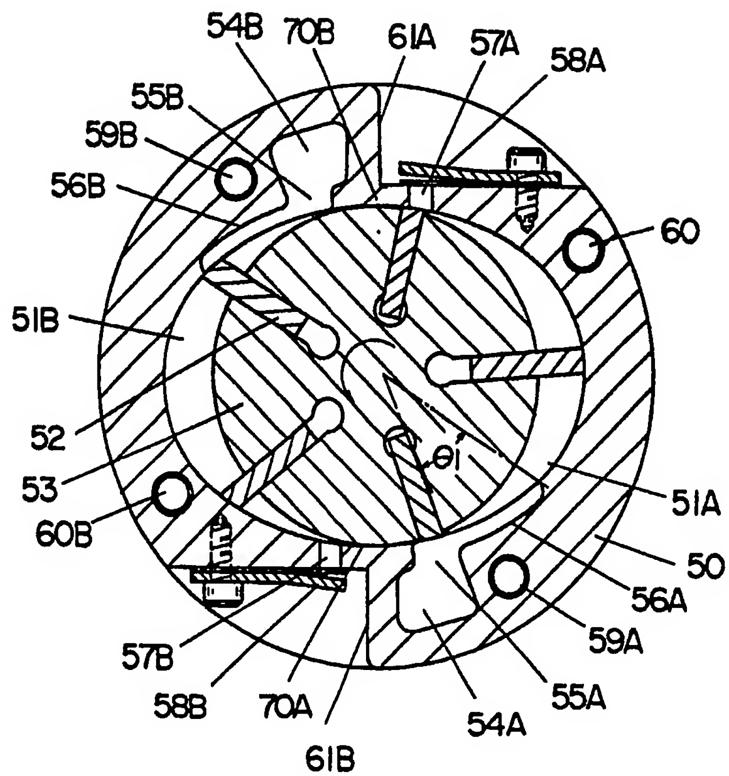

54B
70B
61A
57A
58A
55B
59B
56B
60
51B
52
53
θ1
51A
60B
50
56A
59A
57B
58B
70A
55A
54A
61B

FIG. 4

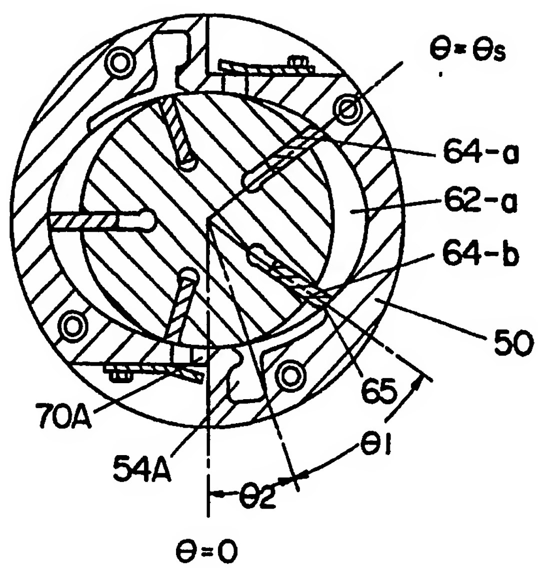
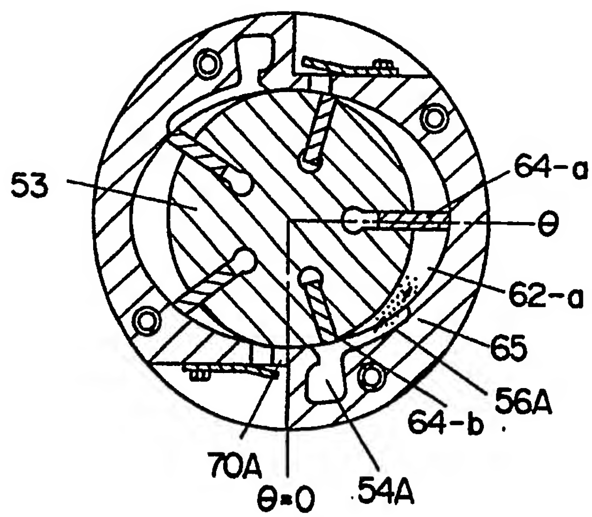


FIG. 4



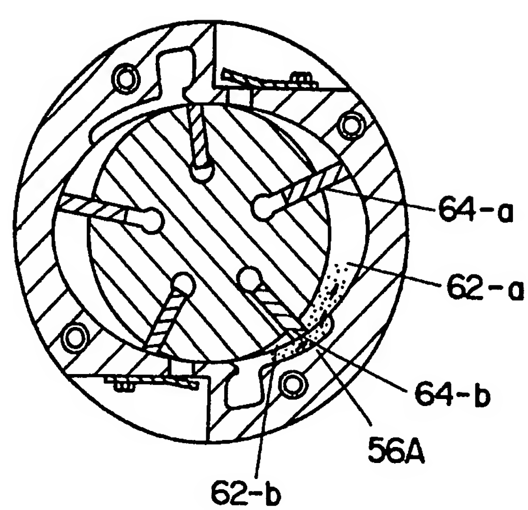


FIG. 5

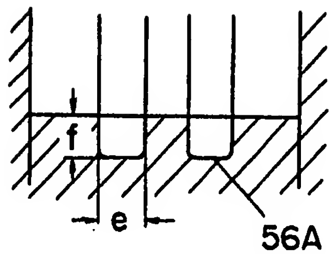


FIG. 6

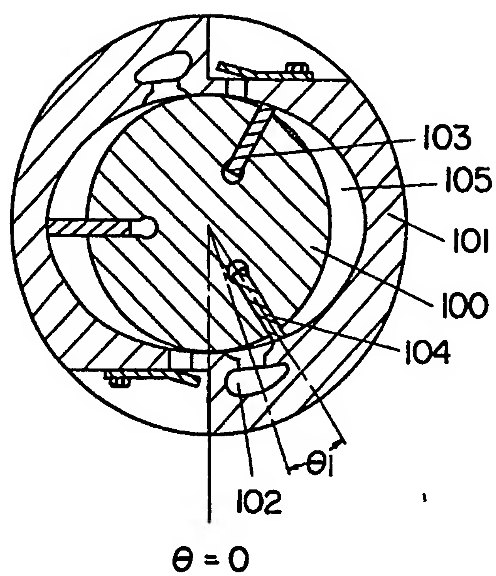
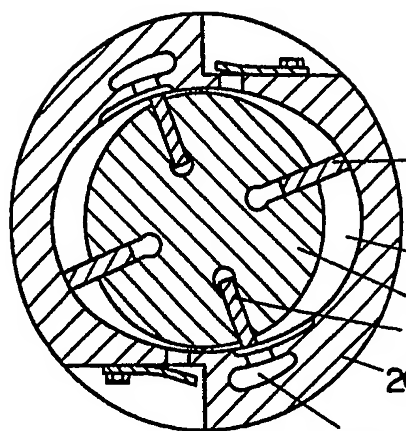


FIG. 7



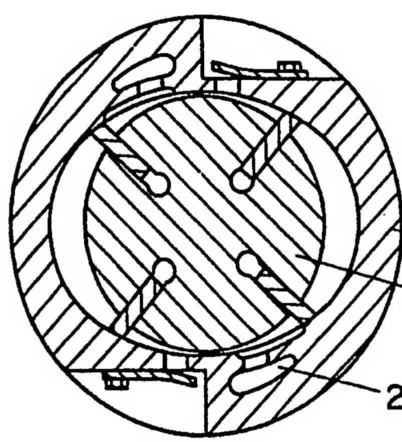


FIG. 8

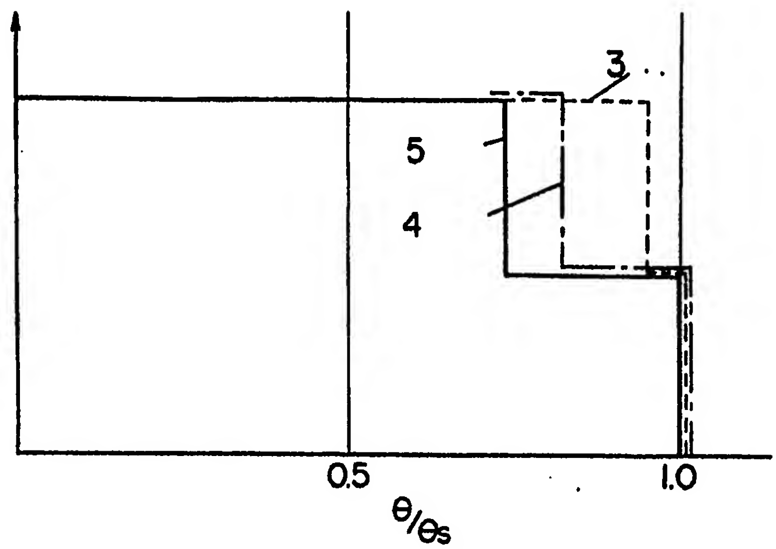

3
5
4
0.5
1.0
θ/θ_s

FIG. 9

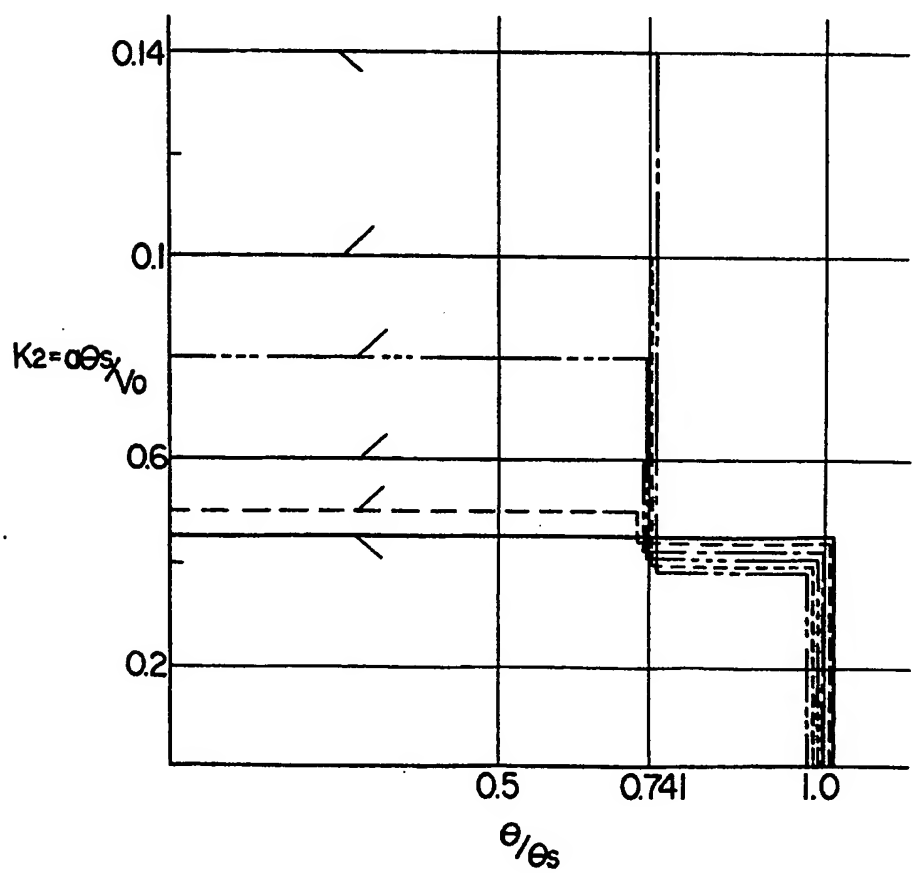

0.14
0.1
$K_2 = a\theta_s/V_0$
0.6
0.2
0.5
0.741
1.0
θ/θ_s

FIG. 10

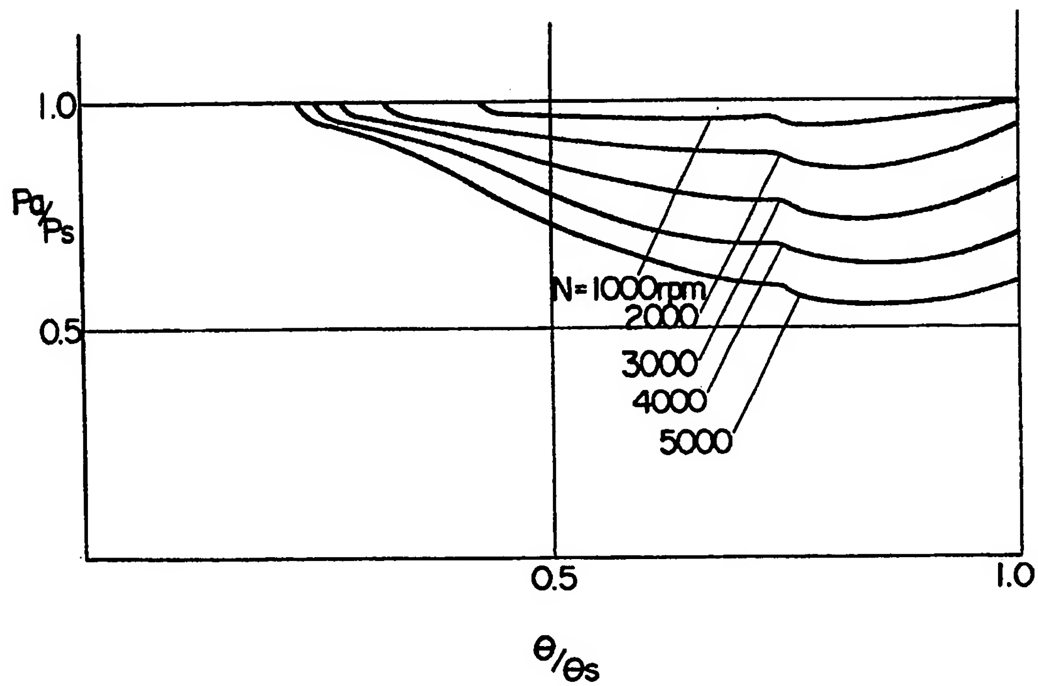


FIG. 11

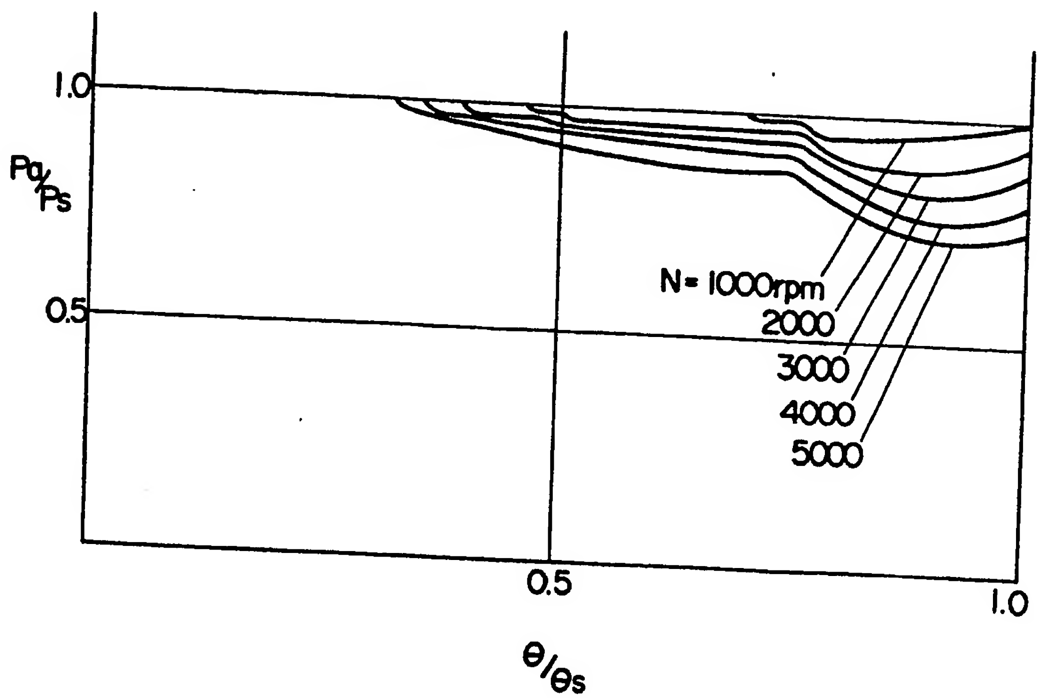


FIG.12

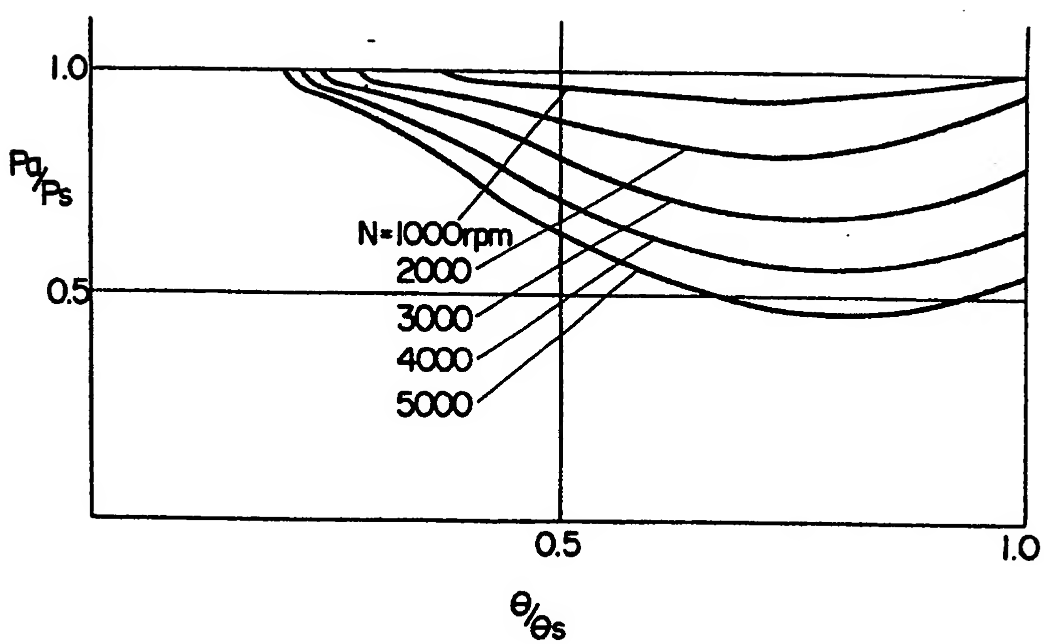


FIG. 13

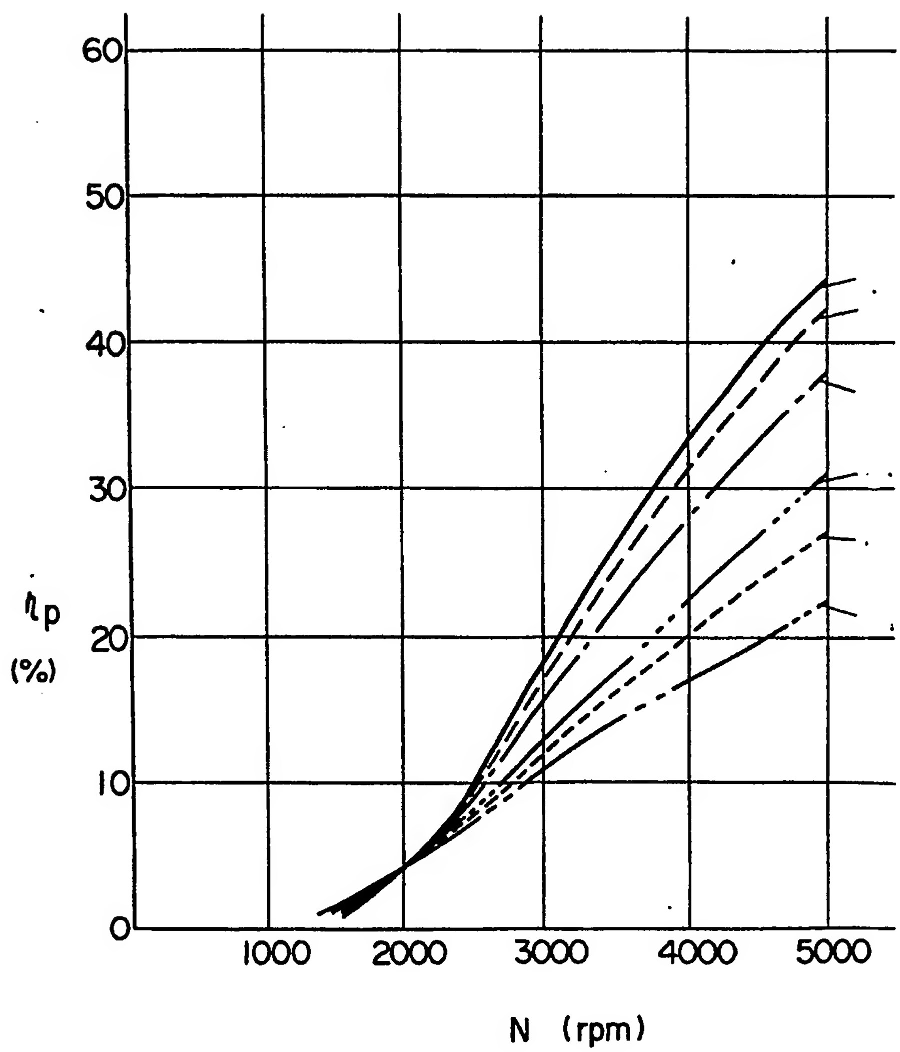


FIG. 14

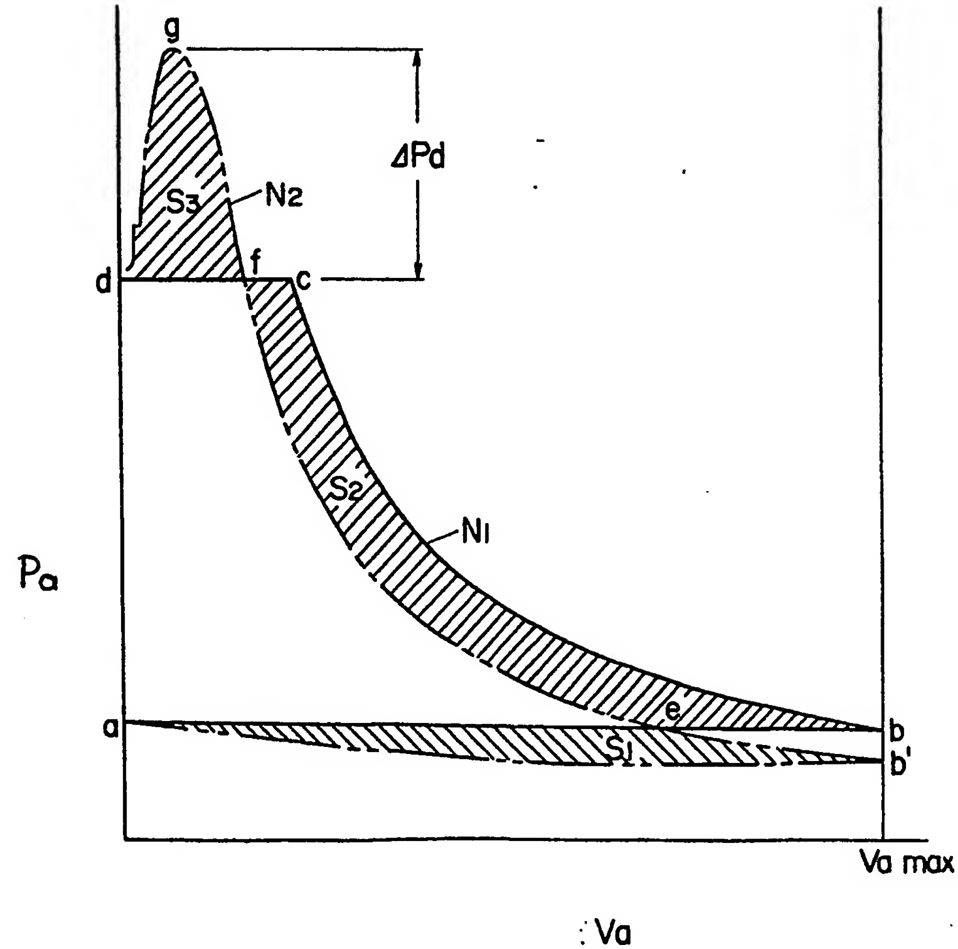


FIG. 15

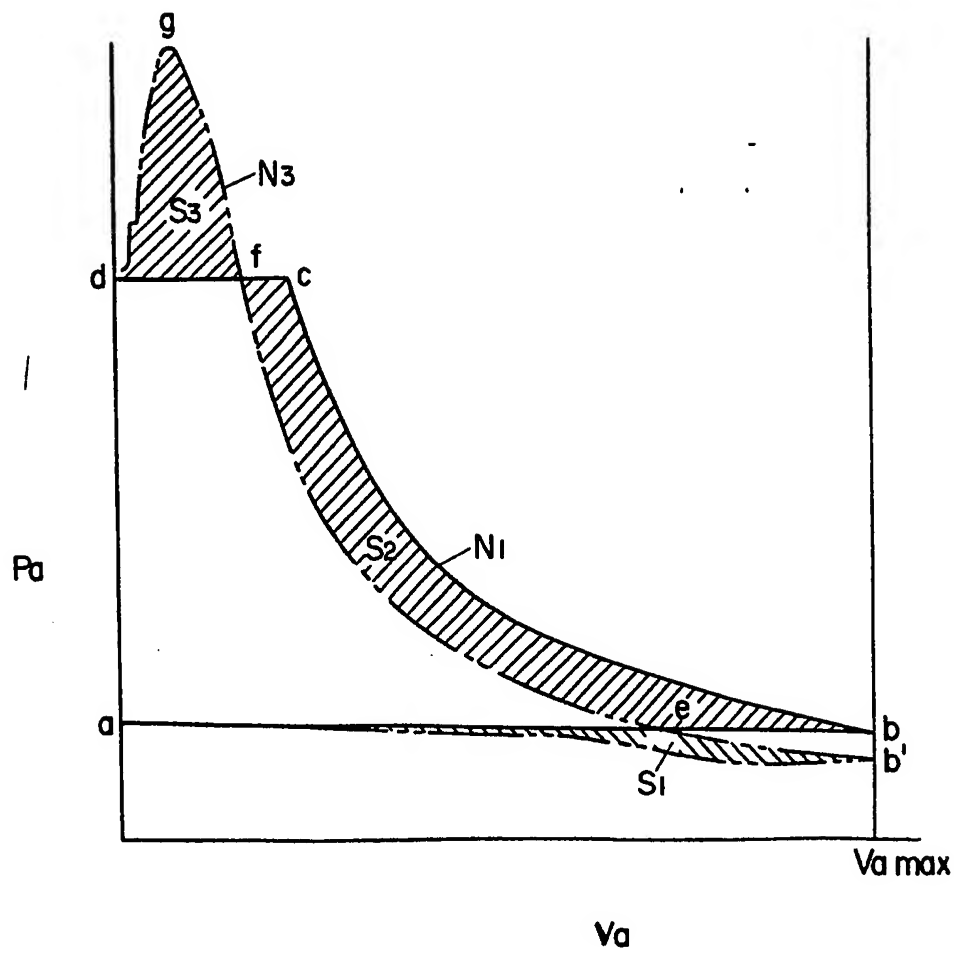


FIG. 16

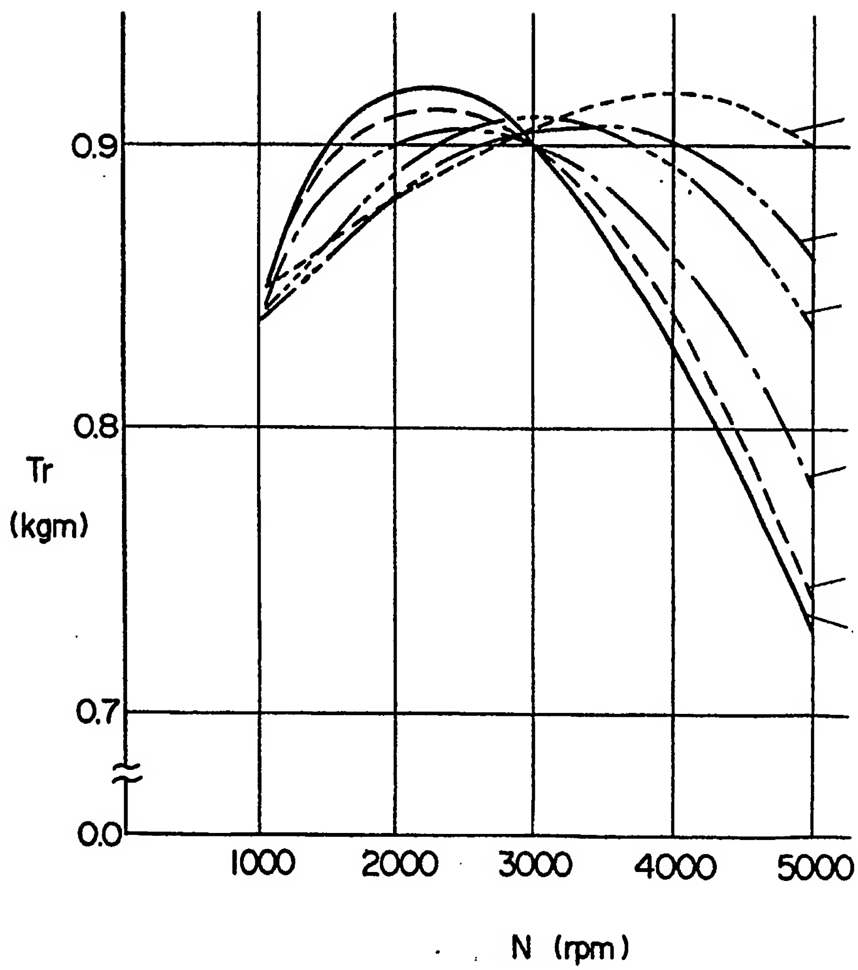


FIG. 17

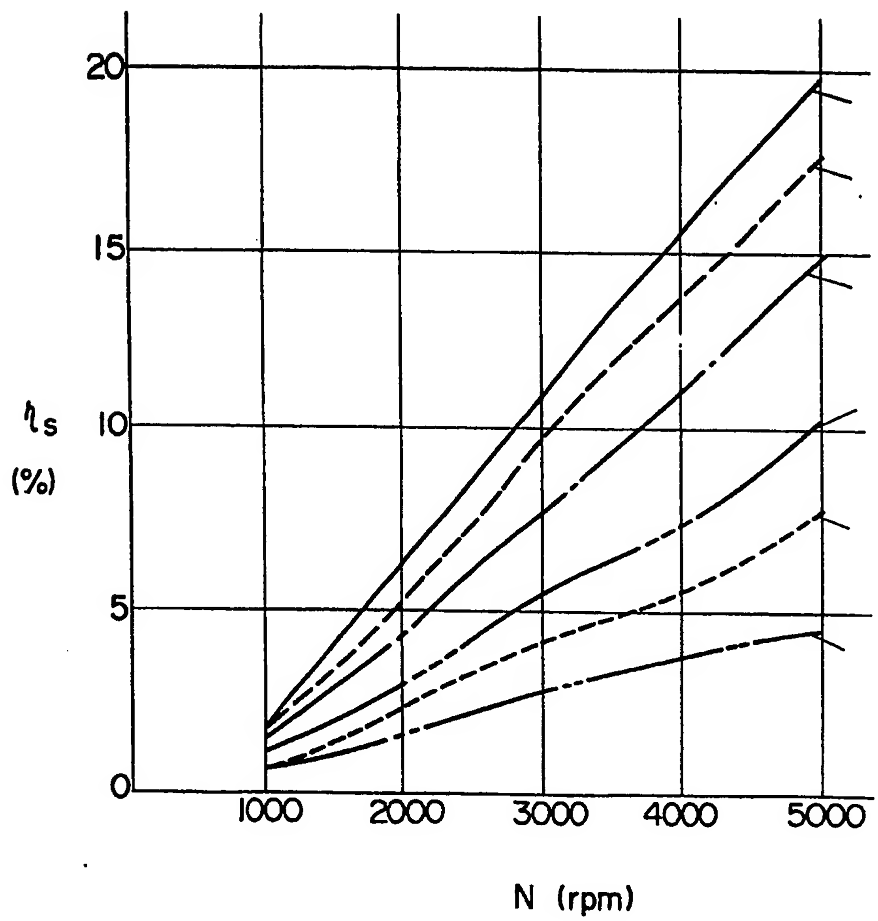


FIG. 18

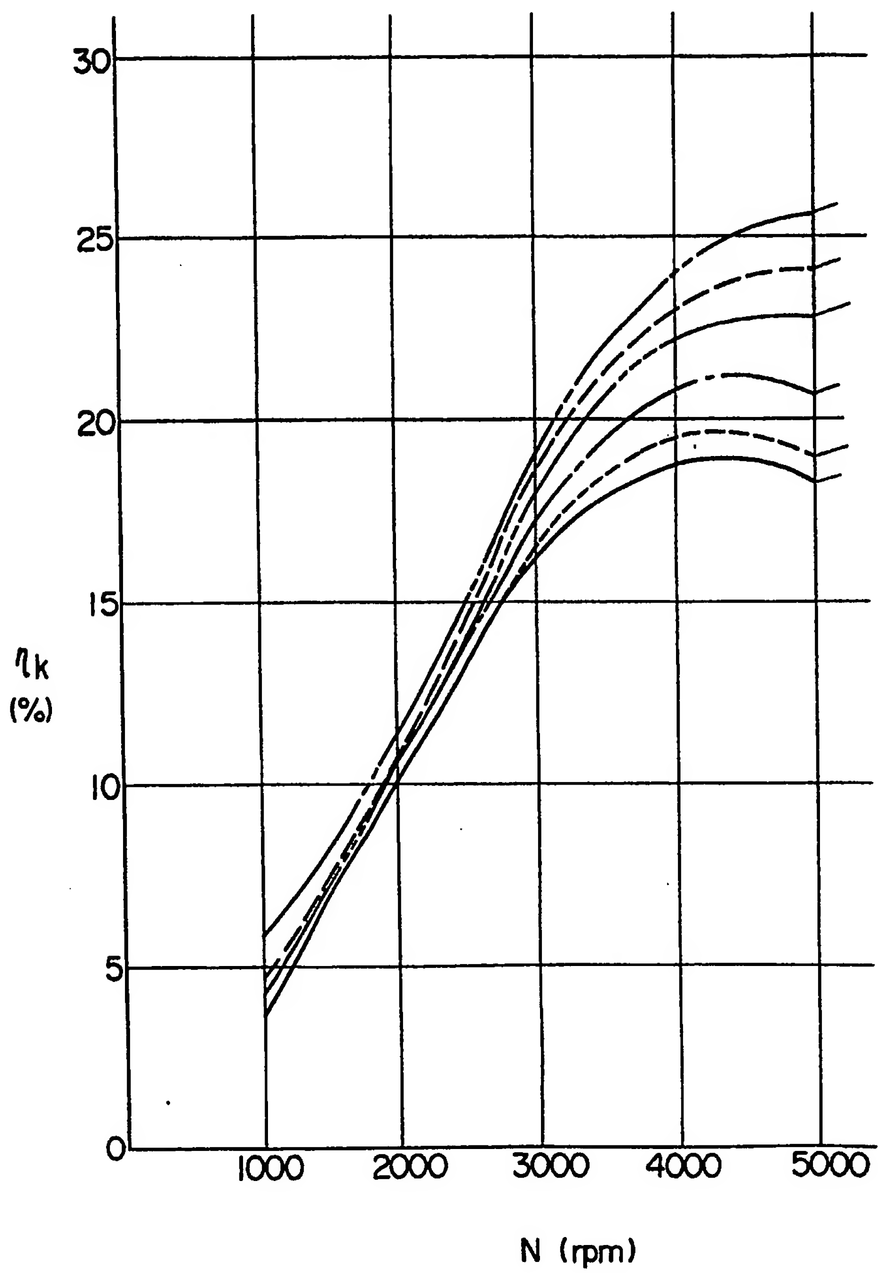


FIG.19

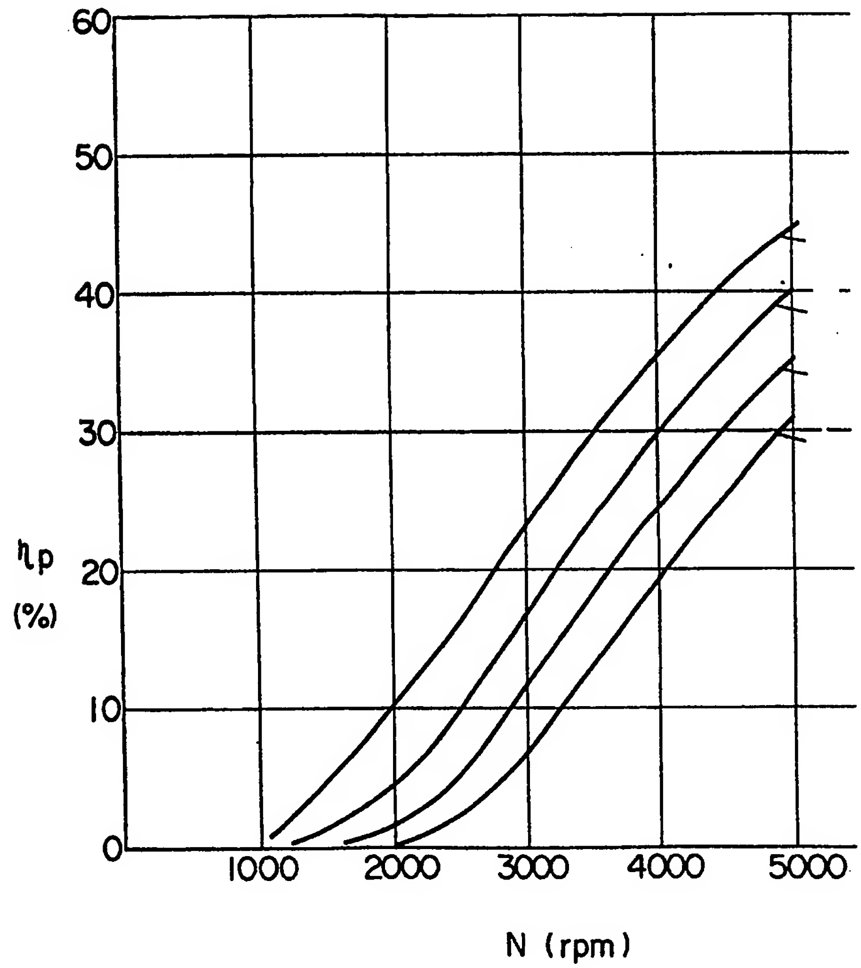


FIG.20

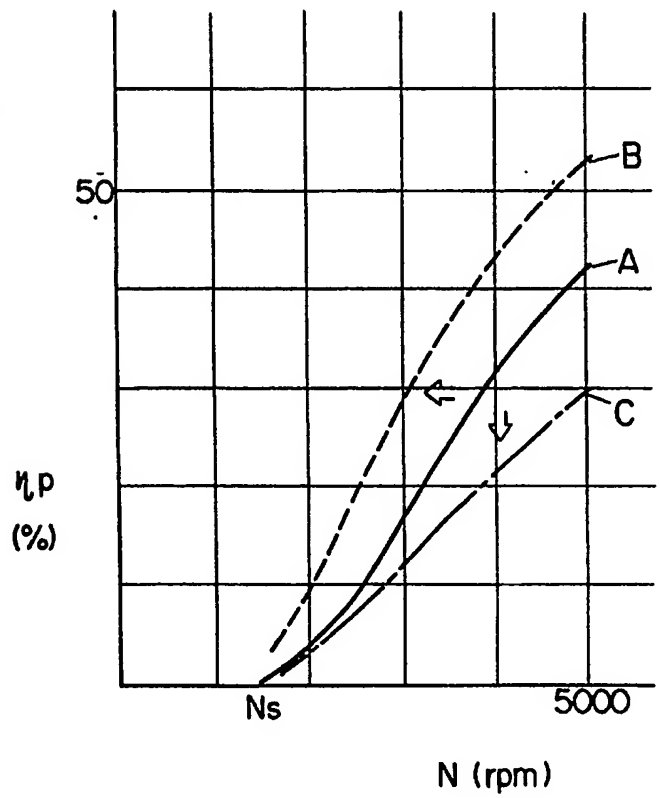


FIG.21

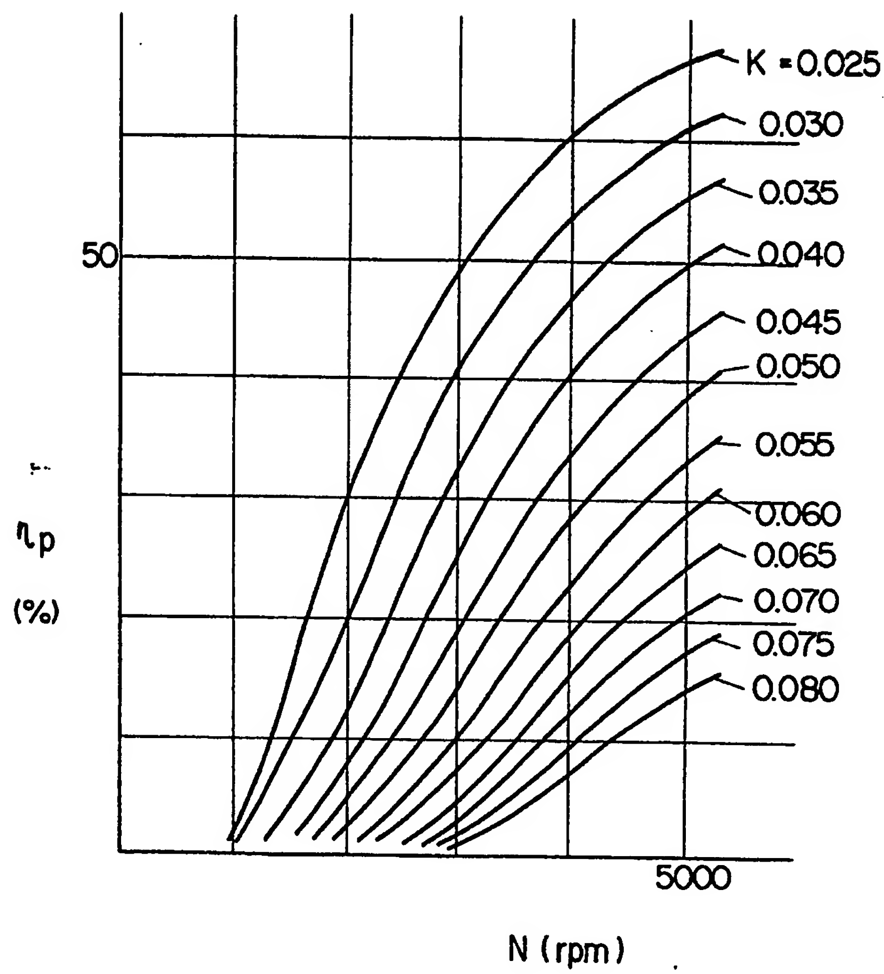
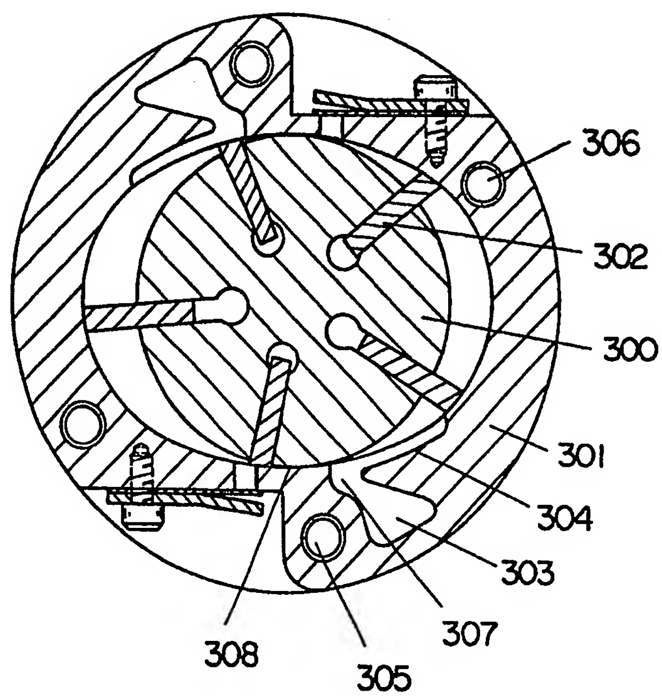

K = 0.025
0.030
0.035
0.040
0.045
0.050
0.055
0.060
0.065
0.070
0.075
0.080
50
η_p
(%)
5000
N (rpm)

FIG. 22



INTERNATIONAL SEARCH REPORT

0101745

International Application No. PCT/JP83/00067

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [3]

According to International Patent Classification (IPC) or to both National Classification and IPC

Int. Cl.^{3} F04C 18/344, 29/08

II. FIELDS SEARCHED

Minimum Documentation Searched [4]

Classification System	Classification Symbols
I P C	F04C 18/344, 29/08, 29/10

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [5]

Jitsuyo Shinan Koho	1926 - 1982
Kokai Jitsuyo Shinan Koho	1971 - 1982

III. DOCUMENTS CONSIDERED TO BE RELEVANT [14]

Category*	Citation of Document, [16] with indication, where appropriate, of the relevant passages [17]	Relevant to Claim No. [18]
A	JP,U, 51-55411 (Diesel Kiki Kabushiki Kaisha) 28. April. 1976 (28.04.76)	1, 2, 3
T	JP,A, 57-126590 (Matsushita Electric Industrial Co., Ltd.), 6. August. 1982 (06.08.82)	1, 2, 3

* Special categories of cited documents: [15]

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"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search [2]	Date of Mailing of this International Search Report [2]
May 13, 1983 (13.05.83)	May 23, 1983 (23.05.83)
International Searching Authority [1]	**Signature of Authorized Officer** [20]
Japanese Patent Office	